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A dynamic organic Rankine cycle using a zeotropic mixture as the working fluid with composition tuning to match changing ambient conditions

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HIGHLIGHTS

- A dynamic ORC using a zeotropic mixture with composition tuning is proposed.
- The working principle is verified theoretically, based on a thermodynamic model.
- Improvements in the resultant power plant's annual power production are analysed.
- The economic benefits have been demonstrated by an economic analysis.

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ABSTRACT

Air-cooled condensers are widely used for Organic Rankine Cycle (ORC) power plants where cooling water is unavailable or too costly, but they are then vulnerable to changing ambient air temperatures especially in continental climates, where the air temperature difference between winter and summer can be over 40 °C. A conventional ORC system using a single component working fluid has to be designed according to the maximum air temperature in summer and thus operates far from optimal design conditions for most of the year, leading to low annual average efficiencies. This research proposes a novel dynamic ORC that uses a binary zeotropic mixture as the working fluid, with mechanisms in place to adjust the mixture composition dynamically during operation in response to changing heat sink conditions, significantly improving the overall efficiency of the plant. The working principle of the dynamic ORC concept is analysed. The case study results show that the annual average thermal efficiency can be improved by up to 23% over a conventional ORC when the heat source is 100 °C, while the evaluated increase of the capital cost is less than 7%. The dynamic ORC power plants are particularly attractive for low temperature applications, delivering shorter payback periods compared to conventional ORC systems.

1. Introduction

The International Energy Outlook 2013 (IEO2013) forecasted that world energy consumption will grow by 56% between 2010 and 2040 [1]. On the other hand, it is estimated that about 20–50% of industrial energy input is discharged as waste heat in the form of hot exhaust gases, cooling water, and heat lost from hot equipment surfaces and heated products. To cut greenhouse gas emissions and increase energy efficiency, it is critical to recover energy from these low-grade waste heat sources. The Organic Rankine Cycle (ORC), which has the same working principle as the normal Rankine cycle, differing only in its use of organic compounds as working fluids, is a feasible solution for the utilisation of such low-grade heat sources. Bianchi and De Pascale investigated different energy systems to convert industrial waste energy to electricity and found that the organic Rankine cycle has the highest thermal efficiency [2]. Yang et al. studied the performance of an ORC for engine exhaust heat recovery and found the thermal efficiency can reach 8.5% [3].

Organic Rankine cycles can also be used for other low-grade heat sources such as geothermal energy [4]. 70% of the global geothermal resource is estimated to exist at temperatures of 100–130 °C [5]. Geothermal energy takes advantage of zero carbon emissions and the sustainable technical potential for Europe is estimated as 350 TW h/yr [6]. Habka and Ajib assessed the perfor-

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mance of some organic Rankine cycle systems using zeotropic mixtures as working fluid for power generation from low-temperature geothermal water, and found that the maximum thermal efficiency can reach 9.01% when the heat source temperature is about 100 °C [7]. Liu et al. analysed the performance of a geothermal organic Rankine cycle using an R600a/R601a mixture as working fluid, and found that it could generate 11% more power than an ORC using pure R600a when the geothermal water temperature is 110 °C [8].

So far, most ORC systems have considered single-pure component organic working fluids [9,10]. The heat transfer efficiency of the evaporator of an ORC is very important. Pinch point temperature difference is often used to analyse the coupling heat transfer in the evaporator. Chen et al. proposed a method to optimise the operating parameters of an ORC with constrained inlet temperature of the heat source, and the pinch point temperature difference in the evaporator [11]. Normally, there is always a mismatch in the temperature profiles between heat transfer fluid and the working fluid because the pure working fluid changes phase at a constant temperature while the heat transfer fluid changes temperature during heat transfer. According to Yu et al., there exists a linear relationship between the integrated-average temperature difference and the exergy destruction in heat exchangers [12]. The integrated-average temperature differences of pure working fluids are large due to the constant temperature during phase change, increasing irreversibility and reducing the cycle efficiency [13]. Zeotropic mixtures with temperature varying during phase change can match the temperature profiles of the heat source and heat sink better. Aghahosseini and Dincer compared the performances of a low-grade heat source for different pure and zeotropic mixture working fluids [14]. Chen et al. analysed a supercritical simple ORC using zeotropic mixture working fluids and found that an improvement of the thermal efficiency can be 10–30% compared to a simple ORC using a pure working fluid [15]. Lecompte et al. examined the thermodynamic performance of a non-superheated subcritical ORC with zeotropic mixtures as working fluids and found that an increase in exergy efficiency in the range of 7.1–14.2% can be obtained compared to pure working fluids [16]. Braimakis et al. studied subcritical and supercritical ORCs based on natural hydrocarbons and their binary mixtures and found that the maximum exergetic efficiencies range from 15% to 40% for heat source temperatures between 150 and 300 °C [17].

An ORC power plant is currently designed by selecting a working fluid to match the heat source and sink temperatures (i.e., T_H and T_L). Xu and Yu's research indicated that the thermal efficiency of a subcritical ORC is related to the critical temperature of the working fluid, and recommended organic fluids with critical temperatures in the range from 20 to 30 K below the heat source temperature to 100 K above it [18]. Considering the heat source and sink temperatures differ from one customer to another, the design options are limited by the availability of suitable organic fluids. Meanwhile, air-cooled condensers are widely used for ORC power plants where cooling water is unavailable or too costly, but they are then vulnerable to changing ambient air temperatures. For instance, the air temperature difference between winter and summer can reach 30 °C in the UK, and can be much higher in locations with continental climates such as Xi'an (China), Warsaw (Poland), and Chicago (US) as shown in Fig. 1. Thus, an ORC power plant designed for a given air temperature (usually the maximum air temperature in summer) operates far from optimal design conditions for most of the year, leading to low annual average efficiencies. This has been identified in literature as a problem facing Organic Rankine Cycles throughout the world [19].

Some of the current ORC systems use multi-component working fluids, but none of them can adjust its composition dynamically. When the ambient temperature varies, these ORC systems either operate at fixed condensation temperatures or have to regulate the mass flow rate of the organic working fluid. However, only regulating the mass flow rate to improve the system performance has limited effectiveness. To address this challenge and fill the knowledge gap, this research proposes a novel dynamic organic Rankine cycle that uses a binary zeotropic mixture as the working fluid and can adjust its mixture composition during operation to match the power cycle with the changing ambient air temperature. A higher annual average thermal efficiency can be potentially achieved by means of the dynamic ORC.

To the best of our knowledge, there currently exist no ORC systems that can dynamically tune their working fluids to match such changing ambient conditions during their operation. In this paper, firstly, the working principle of a dynamic organic Rankine cycle is developed and analysed theoretically. A numerical model is then established and the thermodynamic benefits are analysed. Finally, through a case study, an economic analysis for the resultant dynamic ORC is conducted and compared with a conventional ORC. The results show that the proposed dynamic ORC can improve the power plant's annual power production significantly while the capital cost is only increased slightly.

2. Dynamic ORC concept

It is proposed here that the challenges outlined in the introduction can be addressed by using zeotropic mixtures, which have the following advantages: Firstly, a zeotropic mixture has a variable temperature during phase change. For example, Fig. 2 shows the bubble and dew lines of a zeotropic mixture formed by R245fa and R134a, generated using REFPROP 9.1 [21]. This mixture has previously been investigated in heat transfer research [22]. For a mixture with 70% of low-boiling-point component R134a, the evaporation starts at the temperature T_{bubble} around 279 K and ends at T_{dew} around 290 K. The differential ($T_{bubble} - T_{dew}$) around

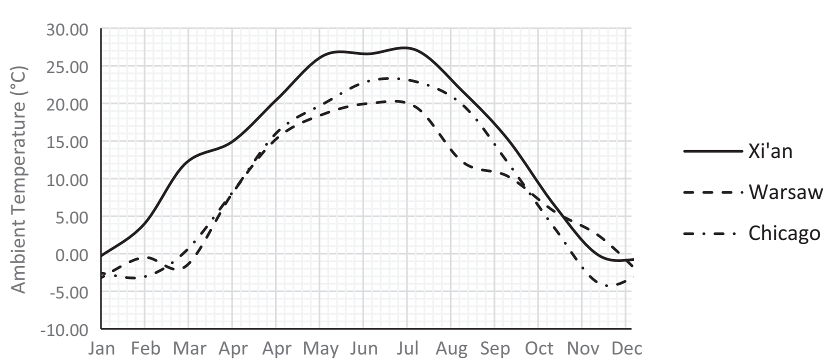

Fig. 1. Average monthly temperatures of three locations with continental climates: Xi'an (China), Warsaw (Poland) and Chicago (US) [20].

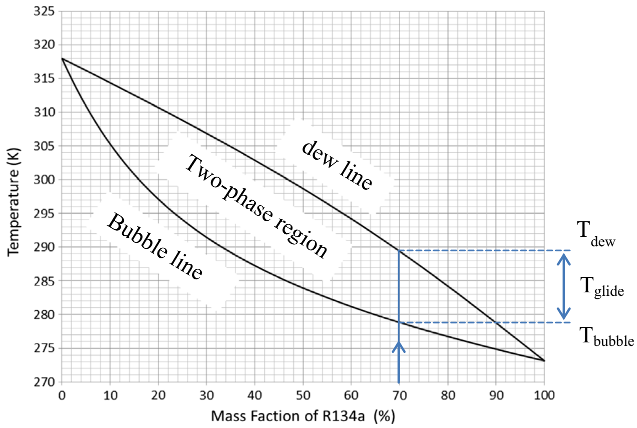

Fig. 2. Dew and bubble lines of a zeotropic mixture of R245fa and R134a at a pressure of 2.9 bar.

11 K is the temperature 'glide' of this mixture, which can be used to match the temperature change of heat transfer fluid in counter flow heat exchangers, so that the irreversibility can be reduced [23–25]. The use of suitable zeotropic mixtures as working fluids can increase cycle efficiency and power production of ORC systems, especially for lower temperature applications (<250 °C) [25]. Secondly, zeotropic mixtures display bubble- and dew-point temperatures between those of the two component fluids, and this varies predictably with mixture percentage. This allows the formation of a mixture with a particular composition to match heat sink temperatures so that more options of working fluids are available [26]. Thirdly, and most interestingly, it is possible to change the composition of the mixture during operation to dynamically match the changing heat source and/or sink temperatures.

This research develops a novel dynamic ORC concept by introducing a composition control mechanism to an ORC operating with a binary zeotropic mixture as the working fluid so that the mixture composition can be dynamically altered during operation to match changing heat sink temperature T_L, ensuring higher efficiency. Such a dynamic composition control has been proposed and analysed for a heat pump system and showed promising results [27]. However, it has never been considered previously for ORC systems according to a comprehensive literature survey and patent search.

Such a Dynamic ORC power plant with an air cooled condenser could be designed and optimised to match the minimum local air temperature in winter as shown in Fig. 3(a). Referring to Fig. 2, the zeotrope will be selected with a high mass fraction of R134a to have a low condensing temperature. As air temperature gradually increases during the seasonal shift from winter to summer, the mass fraction of R134a can be gradually decreased, thus increasing the condensing temperature of the mixture to match the ambient air temperature. As the condensing temperature increases, the evaporating temperature also increases accordingly. In order to ensure the dew point of the mixture is far enough below the heat source temperature to achieve complete evaporation, the evaporating temperature of the mixture can be decreased by reducing the pressure within the evaporator (i.e., reducing both the pump speed and electric load at the generator). After these adjustments, the new cycle is now shown as Fig. 3(b). When approaching the hottest day in the summer, the low boiling point component R134a will be mostly replaced by the high boiling point component R245fa to have a higher condensing temperature. The resultant cycle is shown as Fig. 3(c).

These process are not a one off adjustment but will be carried out with many small steps according to the ambient temperature change as shown in Fig. 1. The process will be reversed as summer gives way to winter. A similar approach can in theory be adopted on a shorter timescale to maximise efficiency over the day–night cycle.

When the heat source and heat sink temperature, T_H and T_L, are given, the Carnot cycle efficiency is defined as

$$\eta_{Carnot} = \frac{T_H - T_L}{T_H} \tag{1}$$

In general, the thermal efficiency of an ORC is proportional to the Carnot cycle efficiency for the given pair of heat source and heat sink temperatures. The higher the differential ($T_H - T_L$), the higher the thermal efficiency for a given T_H. Therefore, a low heat sink temperature T_L is desirable.

As shown in Fig. 4(a), the heat source temperature is fixed as T_H, while the ambient temperature varies over the year with the maximum in summer and the minimum in the winter. In order to demonstrate the performance improvement using dynamic ORC quantitatively, a traditional ORC without composition tuning is selected as a comparative case. Currently, most practical ORC systems are designed based on a fixed condensation temperature that is determined by the designed condensation pressure and the properties of the working fluid. To avoid any leakage of air into the system, the condensation pressure is normally greater than atmospheric pressure. Therefore, the traditional ORC operating with a single working fluid has to be designed according to the maximum air temperature in summer to ensure the working fluid will be fully condensed in the condenser throughout the year. As the season shifts from summer to winter, the ambient temperature decreases. The working fluid in the condenser will be subcooled due to the excessive cooling provided by the colder ambient air

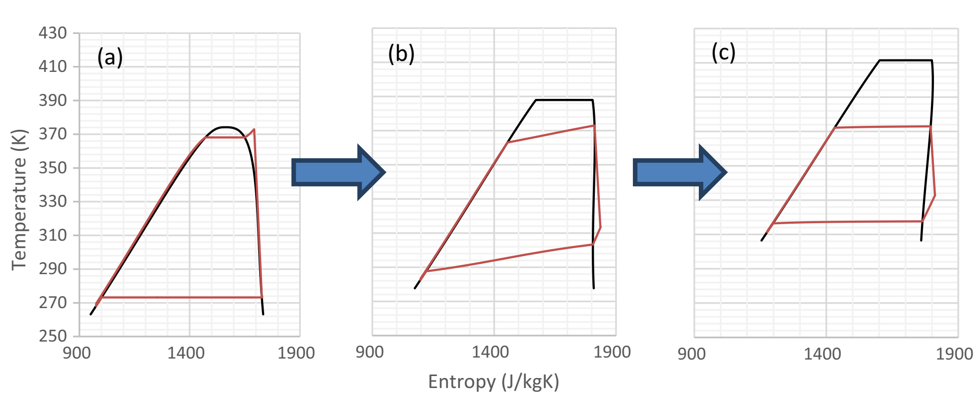

Fig. 3. Change in cycle as the composition of the working fluid is tuned to a changing heat sink temperature, while the heat source temperature remains the same (a) 100% R134a, (b) a 50%:50% mixture of R245fa and R134a, and (c) 100% R245fa.

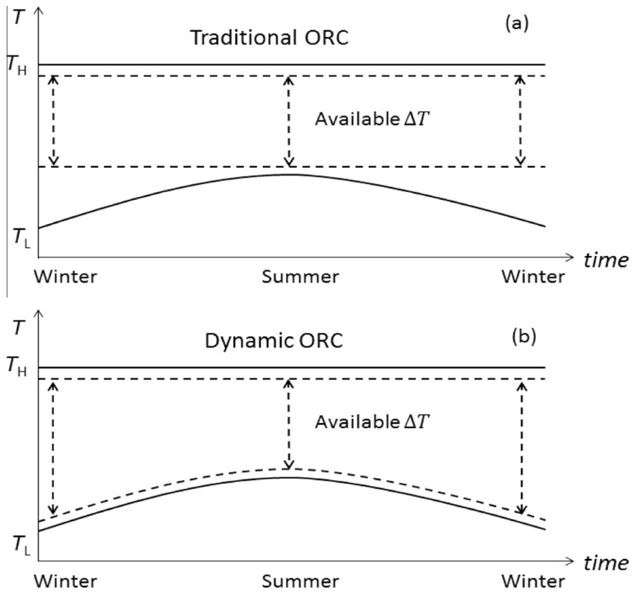

Fig. 4. The comparison between the conventional ORC and the proposed dynamic ORC.

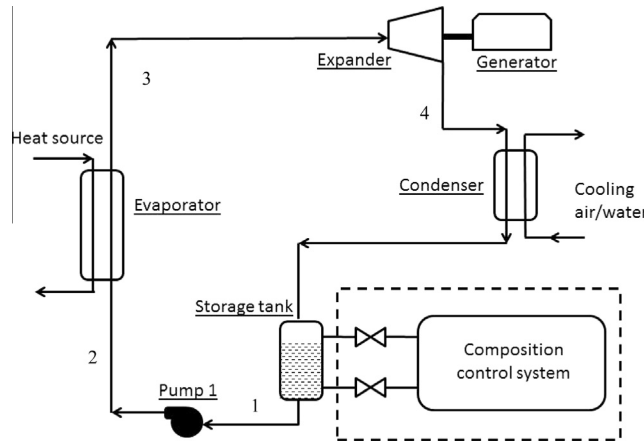

Fig. 5. Schematic diagram of a dynamic ORC power plant.

or water in winter. Such undesired subcooling will increase the heat input at the evaporator and reduce the cycle efficiency. To avoid the undesired subcooling, the flow rate of coolant passing the condenser needs to be reduced to keep the temperature of the working fluid close to the designed condensation temperature. As a result, the available temperature difference (i.e., the differential between the evaporation temperature and condensation temperature) for such a conventional ORC cycle will be kept more or less the same over the year as schematically shown by the dashed line in Fig. 4(a). Therefore, the conventional ORC has a nearly constant efficiency over the year according to Eq. (1).

In contrast, a Dynamic ORC system can change the composition of its working fluid in situ to match the changing ambient air temperature T_L, and thus the available temperature difference varies over the year as shown in Fig. 4(b). The minimum available ΔT appears in summer, which is the same as that for the conventional ORC, while it gets much larger in winter as shown in Fig. 4(b). As such, the proposed dynamic ORC can achieve significantly higher annual average efficiency than a conventional ORC according to Eq. (1).

In the following sections, this paper will numerically demonstrate how the ambient temperature can be matched using this dynamic ORC concept, as well as the benefits of such a Dynamic ORC power plant.

3. Steady state numerical model

As shown in Fig. 5, a conventional ORC power plant has an evaporator (boiler), an expander, a condenser, a feed pump, and a liquid storage tank. In addition to these standard components, the dynamic ORC will have a composition tuning sub system. For the convenience of demonstrating the working principle and benefits of the proposed Dynamic ORC, the composition tuning subsystem can be decoupled from the main power cycle at this stage. It is assumed that the required working fluid composition will be provided to the cycle by the composition adjustment system to match the specified ambient temperature as required, with no other transient or long-term effects. Hence, a standard steady state model can be developed to simulate the ORC with a zeotropic mixture of R134a and R245fa with a series of different compositions which match the specific ambient temperatures. In addition, the turbine is assumed to operate with a constant isentropic efficiency across the range of pressure ratios presented to it, and this assumption is believed to be feasible. In fact, the research from Pierobon et al. [28] suggests that it is possible to maintain a relatively constant isentropic turbine efficiency under a varying pressure ratio by modifying the speed of the turbine. In addition, at high speeds, the isentropic efficiency of the turbine is relatively insensitive to excursions from its operating point.

In order to establish this steady-state model, it is further assumed that there is no pressure or heat loss from piping or heat exchangers, no significant change in velocity, no change in elevation, and no compressibility in the liquid flow. As such, a steady-state numerical model was developed using MATLAB with REFPROP 9.1 providing the thermophysical properties of the working fluid [21].

An Excel file containing values of ambient temperatures for several particular locations around the world was then linked to this numerical code. From this file, the temperature of coolant available for the cycle could be obtained as the heat sink temperature for the power cycle.

The naming convention for points in the cycle is shown in Figs. 5 and 6: The pinch point temperature difference at the condenser outlet is taken to be 5 °C to facilitate the heat transfer from the working fluid to the coolant (i.e., water or air), which is consistent with previous research [23,29]. An additional 2 °C of sub-cooling is also added, to ensure the working fluid is liquid at the pump inlet, giving the following equations:

$$T_1 = T_{ambient} + 5 \tag{2}$$

$$P_1 = P_{sat}(T_1 + 2) \tag{3}$$

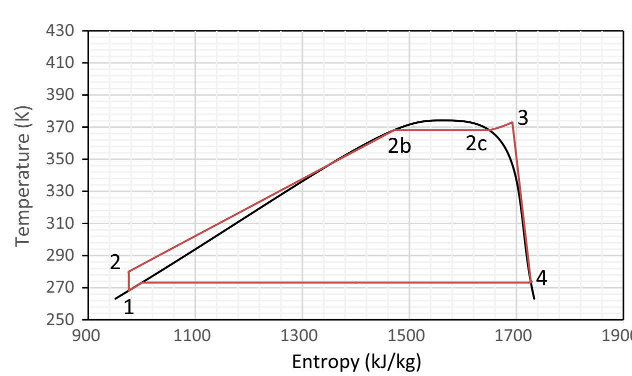

Fig. 6. Temperature–entropy diagram of a conventional ORC cycle.

This allows the condenser pressure of the system to be determined for the hottest day of the year, i.e. when the working fluid is composed entirely of the fluid component with the higher boiling point, in this case R245fa. The condenser pressure is assumed to remain constant throughout the change in composition of the working fluid.

REFPROP 9.1 allows for the calculation of each of the fluid properties on its extensive list given any two other properties, so temperature and pressure information is sufficient to also calculate the enthalpy and entropy of the fluid at this point. Information on the glide curves of the zeotropic mixture of R134a and R245a at this pressure as the composition changes can then be generated, as shown in Fig. 2. Once this information has been obtained, the required working fluid composition to satisfy the conditions of Eq. (2) (a dew point at the condenser pressure 7 °C above the ambient temperature) can be easily obtained for any ambient temperature from the excel file.

Once the required working fluid composition has been established, the remainder of the cycle can be analysed using well-established principles [30,31]. A superheat of 5 °C was applied to reduce the required pressure of the working fluids, and the same 5 °C pinch point temperature difference used in the evaporator, which allows the evaporator pressure to be calculated, using the equations

$$T_3 = T_{heat\,source} - 5 \tag{4}$$

and

$$P_{evap} = P_{sat}(T_3 - 5) \tag{5}$$

To avoid the cycle becoming supercritical, which complicates analysis and comparison, the critical pressure was also calculated. A pressure of 5 kPa below this, or the pressure calculated in Eq. (5) was used as the evaporator pressure, whichever was the lower.

The isentropic efficiency of the pump and the expander were taken to be 90% and 70%, respectively. As discussed above it is feasible to maintain a relatively constant isentropic turbine efficiency under a varying pressure ratio by modifying the speed of the turbine [28]. Assuming isentropic pumping and expansion, h_{2s} and h_{4s} can then be obtained from REFPROP, and used to calculate the actual values, using the equations:

$$\eta_{pump} = \frac{(h_{2s} - h_1)}{(h_2 - h_1)} \tag{6}$$

$$\eta_{expander} = \frac{(h_3 - h_4)}{(h_3 - h_{4s})} \tag{7}$$

Once this has been done, two properties are known for each of the four key points in the cycle; pump outlet, evaporator outlet, expander outlet and condenser outlet, and so Eqs. (8)–(11) can be used to calculate the efficiency of the cycle.

$$W_{pump} = h_2 - h_1 \tag{8}$$

$$W_{expander} = h_3 - h_4 \tag{9}$$

$$Q_{evaporator} = h_3 - h_2 \tag{10}$$

$$\eta_{cycle} = \left(\frac{W_{expander} - W_{pump}}{Q_{evaporator}}\right) \tag{11}$$

This allows for the efficiency of the cycle to be calculated for any ambient temperature provided by the excel spread sheet. By populating this spreadsheet with actual climate data, the year-round performance of a Dynamic Organic Rankine Cycle system can be calculated.

Four key results were calculated by the MATLAB program for each climate dataset. Firstly, the efficiency of the conventional ORC, η_{con}. This is the performance of the Organic Rankine Cycle on the hottest day of the year. Secondly, the efficiency of the dynamic ORC, η_{dyn}. This is the performance of the Organic Rankine Cycle on a given day, given that day's ambient temperature. As the steady-state model is used for these simulations, both η_{con} and η_{dyn} can be calculated using Eq. (11).

Thirdly, the annual average efficiency of the dynamic ORC, $\bar{\eta}_{dyn}$ This is defined as

$$\bar{\eta}_{dyn} = \frac{\sum_1^N \eta_{dyn}}{N} \tag{12}$$

where N is the number of operation days in the year, and is assumed to be 365 here. Finally, ψ, the improvement in annual energy generation, given by

$$\psi = \frac{\bar{\eta}_{dyn} - \eta_{con}}{\eta_{con}} \times 100\% \tag{13}$$

The model was validated against experimental results obtained by Kang [32] by using the same initial parameters, and produced results that were within 2% of his values for all points of the cycle as shown in Table 1. This was considered reasonable in light of the assumptions that had been made in the production of the model.

4. Simulation results of the dynamic ORC

In order to verify the dynamic ORC concept as described above, a selection of case studies were carried out to demonstrate its working principle and benefits. All studies were carried out using the climate data of Beijing, which has a highest temperature of around 38 °C in the summer and a lowest temperature of around −17 °C in the winter [20]. The monthly average temperatures are plotted in Fig. 7(a), and the error bars show the range of fluctuation. The heat source temperature was in the range from 75 °C to 200 °C for these simulations. The obtained simulation results are shown in Figs. 7–9.

Fig. 7(b) shows the required mass fraction of R134a in the zeotropic mixture to match the ambient temperature each day of the year according to Fig. 2 and Eqs. (2) and (3). On the hottest day of the summer, the ambient temperature is about 38 °C, and the

Table 1
Comparison of predictions and experimental data in the Ref. [30].

State point		Present model	Kang's experiments	
1 (Pump inlet)	T (K)	303	303	0.00%
	P (bar)	1.78	1.78	0.00%
	h (kJ/kg)	239.1	239	−0.04%
	s (kJ/kg K)	1.135	1.14	0.40%
2b (Saturated liquid)	T (K)	350.6	350	−0.16%
	P (bar)	7.32	7.32	0.00%
	h (kJ/kg)	305.4	305	−0.13%
	s (kJ/kg K)	1.337	1.34	0.22%
2c (Saturated vapour)	T (K)	350.6	350	−0.16%
	P (bar)	7.32	7.32	0.00%
	h (kJ/kg)	460	460	0.00%
	s (kJ/kg K)	1.778	1.78	0.11%
3 (Expander inlet)	T (K)	353	353	0.00%
	P (bar)	7.32	7.32	0.00%
	h (kJ/kg)	462.9	463	0.02%
	s (kJ/kg K)	1.786	1.79	0.22%
4 (Condenser inlet)	T (K)	318.2	321	0.87%
	P (bar)	1.78	1.78	0.00%
	h (kJ/kg)	441	444	0.68%
	s (kJ/kg K)	1.799	1.75	−2.80%

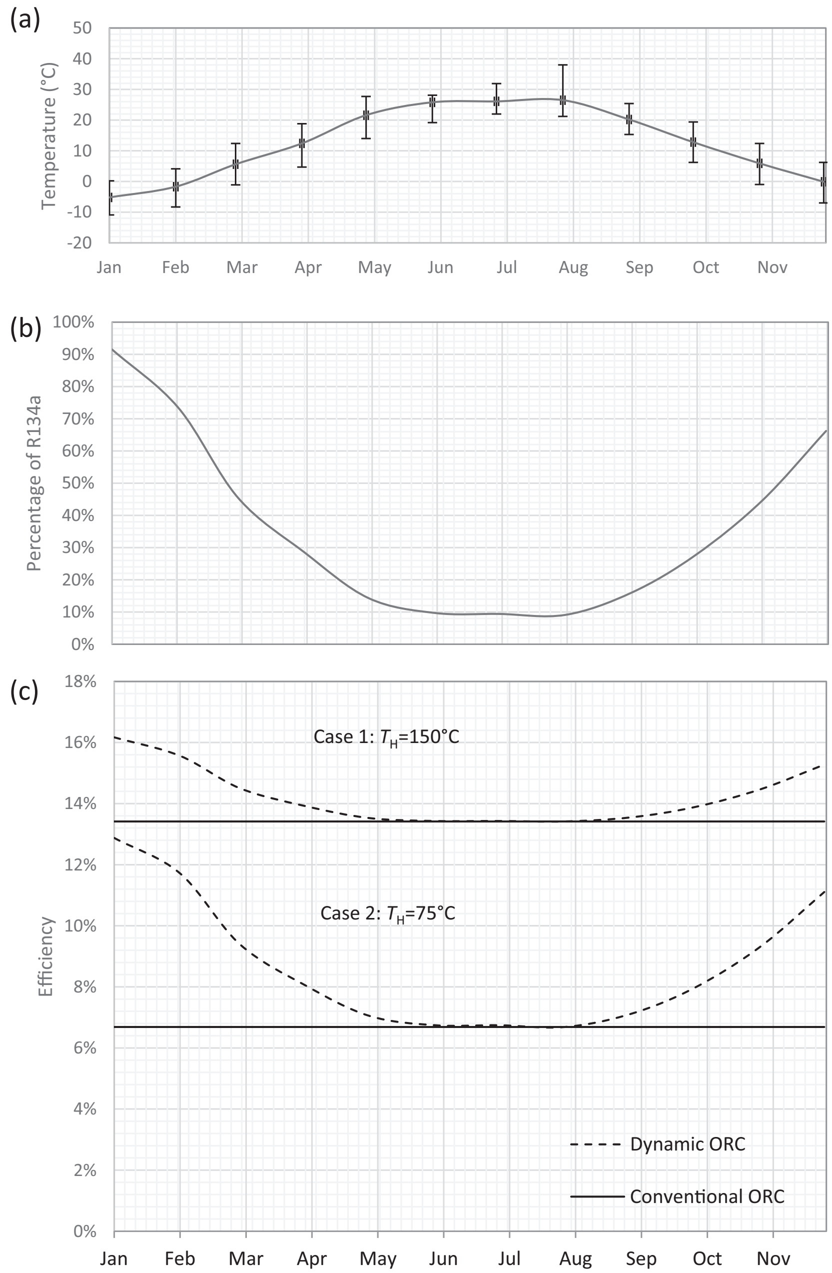

Fig. 7. Comparison between the conventional and Dynamic ORCs, for two different heat source temperatures. (a) Ambient air temperature of Beijing over a year; (b) the required composition of the zeotrope (R134a and R245fa) to match the changing ambient air temperature; (c) the calculated thermal efficiencies for both cycles.

required mass fraction of R134a is about 10% to match this ambient condition. In the coldest day in January, the ambient temperature is around −17 °C. The required mass fraction of R134a increases to around 90%.

Fig. 7(c) shows the annual variation in efficiency of both the conventional ORC and the dynamic ORC over the course of the year, for two different heat source temperatures, 75 and 150 °C. It can be seen that in the colder months, the dynamic cycle is significantly more efficient than a conventional cycle, and also that the effect is greater when the heat source temperature is lower.

Fig. 8 shows the variation in the improvement caused by the dynamic ORC for given climate conditions (i.e., Beijing). It can be seen that the lower the heat source temperature, the greater the potential improvement, due to the low initial efficiency of the system making any gains proportionally more significant.

Fig. 9 shows the effect of the annual temperature variation on the improvement the dynamic cycle can achieve in annual energy production, based on simulation results holding the heat source temperature constant at 100 °C. A clear increase in the benefit of the dynamic cycle can be seen with increasing annual temperature variation.

5. Composition tuning system

Fig. 10 shows the basic principle behind the composition tuning system, which is essentially a distillation column. The temperature decreases up the column, and the blended working fluid is injected half way up. As seen in the glide curve diagram (Fig. 2), at any given temperature, the composition of the vapour and liquid

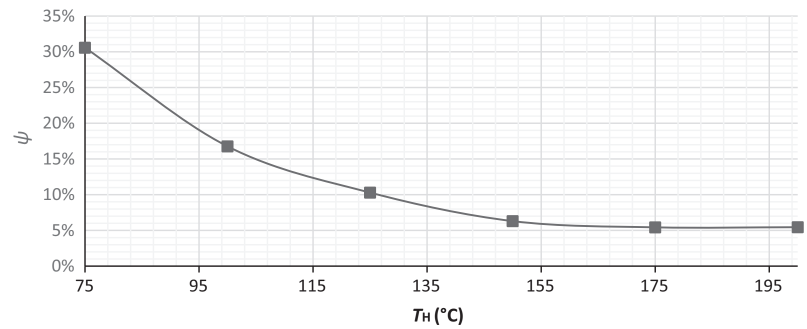

Fig. 8. The effect of heat source temperature on the value of the efficiency improvement ψ. Ambient temperatures for Beijing are used for this simulation.

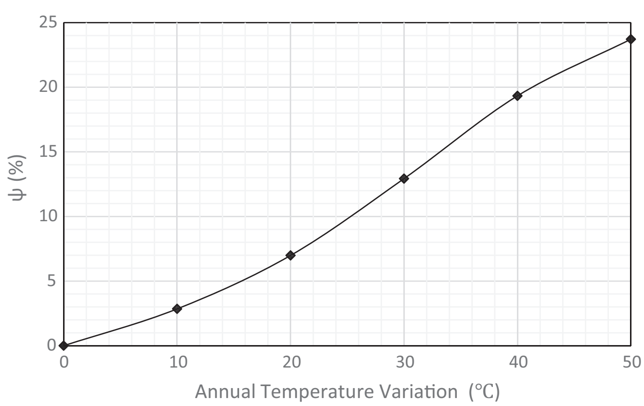

Fig. 9. Effect of annual temperature variation on the efficiency improvement ψ for a heat source temperature of 100 °C.

phases can be obtained. The design of the column is such that the vapour flows upwards due to buoyancy, whereas the liquid rains towards the bottom of the column due to gravity. Each tray in the distillation column exists at a different temperature, with the cooler trays located towards the top, allowing liquid richer in R245fa to rain out even as its concentration in the overall fluid decreases. Eventually, the vapour at the top of the column has been almost completely depleted of R245fa, and the liquid at the bottom has had almost all of the R134a driven out of it, leaving nearly pure liquid R245fa at the bottom of the column, and nearly pure R134a vapour at the top.

The number of theoretical trays needed for distillation can be calculated from Fenske's Equation [33]

$$N_{min} = \frac{\log\left[\left(\frac{x_D}{1-x_D}\right)\left(\frac{1-x_B}{x_B}\right)\right]}{\log \alpha_{ave}} \tag{14}$$

where x_D is the proportion of the more volatile component in the distillate, x_B is the proportion of the more volatile component in the bottoms, and α is the relative volatility of the feed. This result can, as a rule of thumb, be doubled to give the actual number of trays required [33].

The number of trays against the purity of product the resulting distillation column will produce is plotted below in Fig. 11, showing the increase in purity of the product caused by increasing the number of trays in the distillation column. Its implementation in our Dynamic ORC power plant would be as shown in Fig. 12. The liquid R245fa collects at the bottom of the distillation column, where it can be pumped to a storage tank. The system is designed such that R245fa can be stored as a liquid at condenser pressure, so no extra pressurization is needed. The R134a vapour collects at the top of the system. A compressor then increases the pressure of the vapour to a level at which it can also be stored as a liquid at the ambient temperature. This causes an increase in the temperature of the vapour, so it must be passed through a post-cooler before collecting in the storage tank. The working fluid removed from the power generation cycle can be replaced by simply opening a valve and pumping the desired component back into the system.

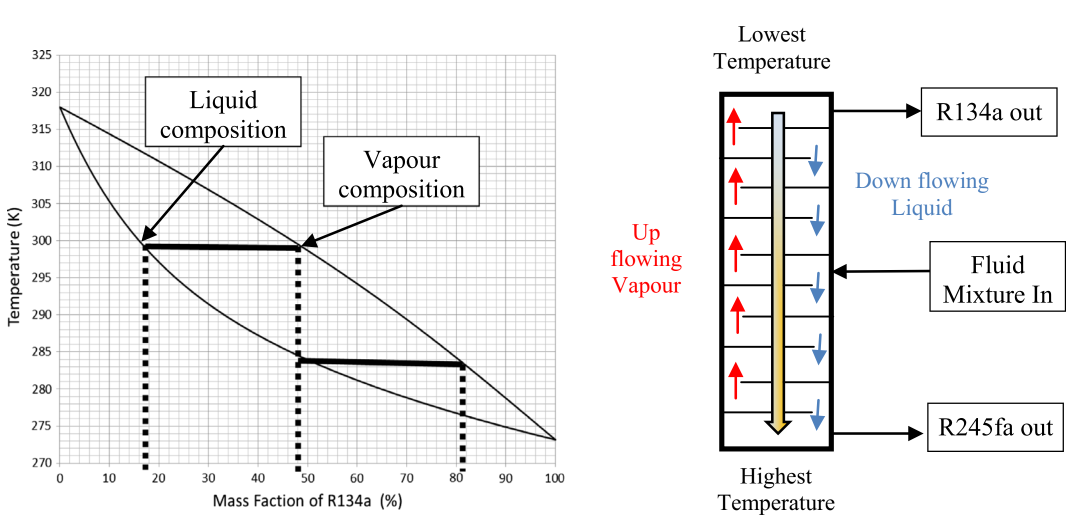

Fig. 10. Distillation process for separating the components of the zeotropic fluid used in this research.

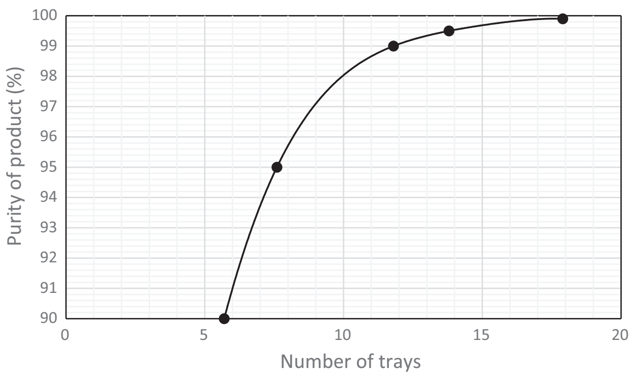

Fig. 11. Effect of number of trays on product purity.

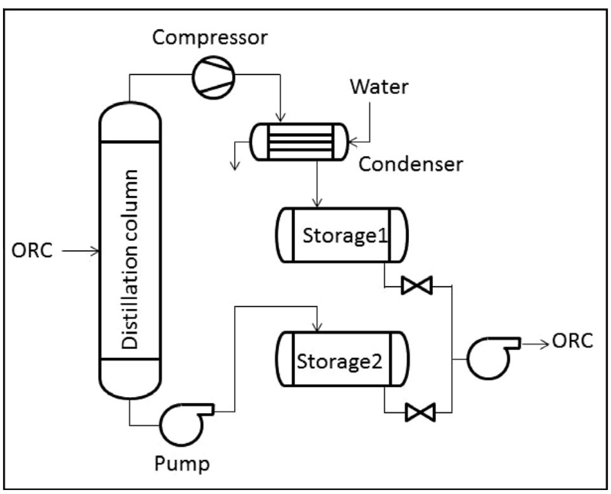

Fig. 12. Schematic diagram of the composition tuning sub-system.

Refrigeration and heat pump systems generally have a refrigerant charge of 0.25–0.5 kg/kW installed capacity [34]. This power figure refers to the amount of heat the heat pump can transfer, which corresponds to the evaporator and condenser loading in an ORC system. As per the simulations used in our case study, the efficiency for an ORC with a heat source temperature of 100 °C operating under Beijing's ambient conditions has an efficiency of 9.89% meaning that the power transferred through the evaporator and condenser is taken to be between 10.11 and 9.11 times the energy extracted through the expander, respectively. This implies that Organic Rankine Cycles will have a system charge of 2.278–5.05 kg/kW.

The charge of working fluid for a 1 MWe plant would therefore be estimated in the range of 2778–5050 kg. This paper assumes a system charge of 4000 kg. The two greatest sources of energy required for the composition tuning system are the energy required to vaporise the working fluid in the distillation column, and the power required to compress the distilled R134a to storage pressure.

Therefore, the amount of energy required to switch out the working fluid can be given by Eq. (15)

$$E_{composition\ tuning} = E_{column} + E_{compressor} \tag{15}$$

The heat required for complete separation of a binary mixture is given by Stichlmair and Stemmer [35] to be

$$\dot{Q}_{min} = \dot{F}r\left(\frac{1}{\alpha - 1} + x_{Fa}\right) \tag{16}$$

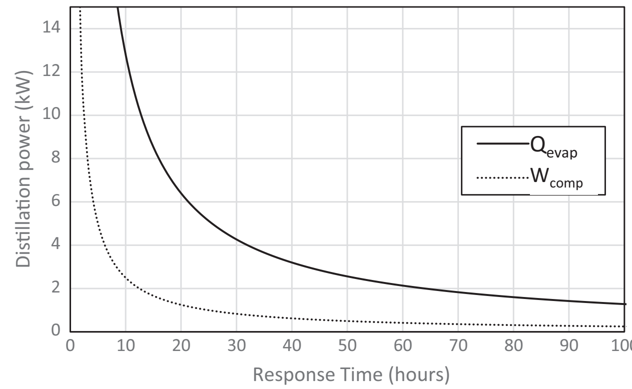

Fig. 13. Power required for distillation and compression against the time to perform it.

where $\dot{F}$ is the feed flow rate, r is the heat of vaporisation of the feed, x_{Fa} is the fraction of lighter component in the feed, and α is the relative volatility, or the ratio of vapour pressures of the two components. This theoretical value of Q_{min} is estimated by Stichlmair and Stemmer to be about 90% that of a practical system.

This will give the energy required for complete separation of a given composition of feed. The energy required to replace the entire charge of an ORC system can be obtained by dividing the process into 100 steps, and summing them as

$$Q_{total} = \sum_{i=0}^{100} Q_i \tag{17}$$

In this way, the total energy required to distil the entire charge of the system was calculated using MATLAB to be 613 MJ, giving an average annual distillation power of 19.42 W.

In order to compress 1 kg of R134a vapour from condenser pressure (2.9 bar) to storage pressure (15 bar to ensure liquid condition at 40 °C) requires 30 kJ of energy [20]. Over the course of a year, the entire charge of the system needs to be removed and compressed to storage pressure, so the total energy requirement for the compressor is 120 MJ. Averaged over the course of the year, this leads to a parasitic compressor power of 3.8 W. This results in a total average energy draw for the distillation system of 23.22 W year-round.

However, in the event of extreme weather conditions, such as cold snaps or heat waves, it is desirable to have the capacity to completely replace the working fluid in a short period of time, e.g., 12 h for the worst case scenario. This results in a compressor power of 2.8 kW, and a distillation column heat demand of 14.26 kW, according to the equations:

$$\dot{W}_{comp,installed} = \frac{365 \times 24}{12} \dot{W}_{comp,average} \tag{18}$$

$$\dot{Q}_{column,installed} = \frac{365 \times 24}{12} \dot{Q}_{column,average} \tag{19}$$

Fig. 13 shows the required power for the distillation column and the compressor with a varying response time of the system. It can be seen that the shorter the response time, the larger the required capacity of the distillation system. This data is used in the following section to perform an economic analysis of the system.

6. Economic analysis

An economic analysis of the system was carried out, focusing on the case study of a 1 MW power plant with the climate conditions

Table 2
Coefficients for component pricing [31].

	Pump	Evaporator	Expander	Condenser	Generator
K_1	3.3892	4.3147	2.2476	4.0336	
K_2	0.0536	−0.3030	1.4965	0.2341	
K_3	0.1538	0.1634	−0.1618	0.0497	
X	40 kW	2000 m^2	1000 kW	2000 m^2	1000 kW
C_1	−0.3935	0.03881			
C_2	0.3957	−0.11272			
C_3	−0.00226	0.08183			
B_1	1.89	1.63	3.5	1.96	
B_2	1.35	1.66		1.21	
F_m	1.5	1		1	
F_{BM}			3.5		1.5

in Beijing, and considering the sizing of the critical components, the extra working fluid, and the extra energy consumption to run the composition tuning system. This was used to calculate the Payback Period and Net Present Value of a case study of a system utilising a 100 °C heat source.

The cost of each major component was calculated using the Chemical Engineering Component Price Index, as utilised by Wang et al. [31], and outlined in the following equations

$$\log Z_p = K_1 + (K_2 \log X) + (K_3(\log X)^2) \tag{20}$$

where Z_p is the component price in US Dollars, K_1, K_2 and K_3 are constants as defined in Table 2, and B is a characteristic of the component under analysis, this being power for fluid machinery and surface area for heat exchangers, which was taken to be 2 m^2/kW, as per Zhang et al. [36].

The generator cost is calculated using the following equation [31]

$$Z_p = \frac{1,850,000 W_{expander}}{11,800^{0.94}} \tag{21}$$

These component prices assume that they are made from carbon steel, and operate at ambient pressure. If they are made of other materials, or operate at other pressures, a correction factor must be applied, according to the following equations.

$$Z_{BM} = Z_P F_{BM} \tag{22}$$

$$F_{BM} = B_1 + (B_2 F_M F_P) \tag{23}$$

where Z_{BM} is the corrected cost, F_{BM} is an overall correction factor, B_1 and B_2 are correction factors given in, and F_M and F_P are correction factors for material and operating pressure respectively. Where a value of F_P is not directly given, it can be calculated using the equation [31]

$$\log F_p = C_1 + (C_2 \log P) + (C_3(\log P)^2) \tag{24}$$

where P is the operating pressure of the component.

Current market prices of R245a are roughly £25/kg at the time of writing this paper, giving a cost of £100,000 for the 4000 kg required for a 1 MW system.

Using Eqs. (20)–(24), and an exchange rate of £0.66 to the US dollar, the cost of the components for a conventional 1 MW ORC system is estimated as follows in Table 3:

Quoilin et al. [37] have collated data on real-world ORC Systems, plotting the price per kW against the system size, for fully-commissioned systems, and for the module costs only. Their findings show that the module-only cost for systems in the 1 MW range is €1000–3000 per kW (equivalent to £750–2250). The figure generated by the above analysis falls right in the middle of this range, which suggests it is a reasonable estimation.

The predictions of this method do not hold for smaller components, so an approximation had to be made to estimate the cost of

Table 3
Component prices for the conventional ORC power plant.

Component	Cost	%
Pump	£298,915	17.96
Evaporator	£84,056	5.05
Expander	£441,075	26.49
Condenser	£468,302	28.13
Working fluid	£100,000	6.01
Generator	£272,419	16.36
Total	£1,664,767	100

the composition tuning system. As the distillation system consists of heat exchangers, pumps and turbo-machinery, an assumption was made that the component costs were of the same order as that of an ORC system of comparable size. This allowed the system data assembled by Quoilin et al. [37] to be used estimate a cost for the composition tuning system.

Section 5 calculated the worst case scenario of having to replace the entire charge of the system in 12 h, giving a required compressor capacity of 2.1 kW, and a heat requirement for the distillation column of 14.6 kW.

Quoilin et al.'s data shows that ORC systems in this power range have a cost of €2000–8000/kWe, which is equivalent to £1500–6000/kWe. If we assume an average thermal efficiency of 10%, this translates to a cost of £150–600/kW thermal energy. We can therefore roughly estimate the cost of the composition tuning system as £150–600/kW thermal energy. For the worst case scenario, the cost of the composition tuning system is estimated as £1600–6400, but this cost drops off sharply with an increasing response time of the system, as shown in Fig. 14, stabilising at a maximum of about £800 for longer timescales, such as would be experienced if the composition tuning system were operated for a short period each day to keep pace with seasonal temperature shifts.

In order to successfully operate a dynamic cycle such as the one described in this paper, the amount of working fluid in the system would need to be doubled compared to a conventional ORC of the same installed capacity. On the hottest day of the year, the working fluid in the power cycle is 100% R245fa. An equal mass of R134a is required in storage to completely replace this for the coldest day of the year. As discussed above, this would increase the installation cost of the system by £100,000, or 6.01%. In theory, this could be reduced by using component fluids with a greater difference between their boiling points, and therefore avoiding the need to use the full range of compositions to match the variation in temperature.

Table 4 shows that the comparison of the costs between a dynamic ORC system and a conventional ORC power plant. The

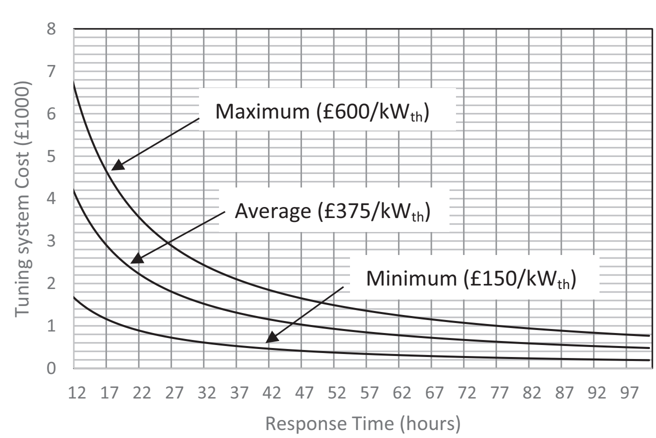

Fig. 14. Tuning system cost against desired response time.

Table 4
Economic analysis of the dynamic ORC power plant.

	Conventional ORC	Composition tuning system	Extra working fluid	Dynamic ORC
Cost	£1,639,767	£6400	£100,000	£1,746,167
%	100	+0.39	+6.01	106.4

composition tuning system will introduce around 0.39% of costs, while the extra working fluid will introduce about 6.01% of the costs. Overall, the total costs of a dynamic ORC power plant is projected to be 106.4% that of a conventional system of the same capacity. This increased cost will be offset by a greater energy production.

Quoilin et al. [37] have collated the full-plant prices of several existing ORC plants which vary from €2000 to €4000/kWe (equivalent to £1500 to £3000/kWe) installed capacity for systems with a capacity over 1 MWe. For the following economic calculations this paper will use an intermediate value of £2500/kWe for our case study using a 100 °C heat source, which accounts for additional installation and commissioning costs over and above the module costs calculated earlier and presented in Table 4. Higher temperature heat sources will tend to lead to greater efficiency, and reduce the cost per kW of the system due to the smaller relative size of components, especially the heat exchangers.

The price of electricity was taken to remain constant at £0.05/kW h in the UK. The discount rate, which reflects the current value of future energy generation, was taken to be 10%, which is in agreement with other small- to medium-scale energy generation projects [38,39]. This allowed the Net Present Value of the conventional and dynamic concepts to be compared over time, according to the equation

$$\mathrm{NPV} = -C + \sum_{N=0}^{Years} \frac{R_N}{(1+d)^N} \tag{25}$$

where N is the number of years into the future, R_N is the annual revenue from the system, d is the discount rate, and C is the capital employed, or the total initial investment in the ORC system. This metric is considered by Lecompte et al. to be the best for comparison of similar systems, although it does make assumptions that are dependent on time and location [40]. Systems can also be compared on the basis of Return on Capital Employed

$$\mathrm{ROCE} = \frac{\sum_0^N R_N}{C} \tag{26}$$

where Capital Employed is the initial investment in the ORC system.

Figure 15 shows the net present value of both the dynamic ORC and conventional ORC power plants. The heat source temperature is given as 100 °C, while different climate conditions (i.e., the annual temperature variations) have been used to achieve different values of efficiency improvement ψ. Moreover, for the convenience of comparison, the summer temperature is kept as the same for all cases, while the winter temperature varies. As such, the effect of the value of ψ on the power plants' payback period can be compared. The comparison clearly shows that the dynamic ORC power plant has the higher NPV. Furthermore, the higher the value of efficiency improvement ψ, the higher the NPV.

Extracting the data from Fig. 15, Table 5 shows the change in NPV for varying values of efficiency improvement ψ in detail. The conventional ORC has a payback period of about 114 months. This is within the bounds of the payback periods calculated by Wang et al. [31], whose analysis suggested payback periods between 60 and 140 months, depending on the particulars of the cycle. The dynamic cycle has a shorter payback period, which decreases with increasing ψ. With a value of ψ of 20%, the payback period is around 19.3% shorter than for the conventional cycle. The dynamic cycle has a higher NPV than the conventional cycle for all plant lifetimes greater than 3 years.

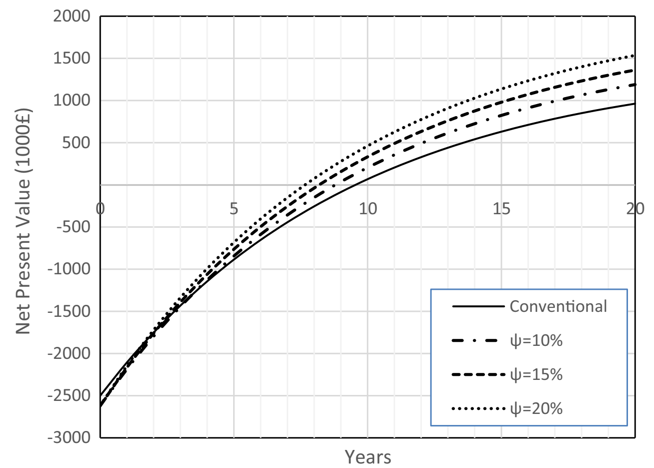

Fig. 15. NPV over time for the conventional cycle and three different values of ψ (with a capital cost of £2500/kW).

Table 5
Comparison of economic data for differing values of ψ (£2500/kW).

	Payback period (months)	NPV (20 years)	ROCE
Conventional cycle	114	£963,000	1.39
Dynamic ORC: ψ = 10%	106	£1,189,000	1.45
Dynamic ORC: ψ = 15%	97	£1,362,000	1.52
Dynamic ORC: ψ = 20%	92	£1,545,000	1.59

7. Conclusions

This paper has proposed a novel Dynamic ORC concept, which uses a binary zeotropic mixture as the working fluid and can dynamically alter mixture composition to match the ambient air temperature, so that the resultant power plant can achieve higher annual average efficiency than a conventional ORC using a single component working fluid. A numerical model based on the thermophysical properties provided by REFPROP 9.1 has been developed to simulate the proposed Dynamic ORC. The simulation results have demonstrated that a dynamic ORC can improve annual energy generation in a continental climate such as Beijing by over 20% when the heat source temperature is of the order of 100 °C. Furthermore, as the heat source temperature decreases, the benefits of Dynamic ORC become more significant, showing that the Dynamic ORC has great potential for low temperature applications.

An economic analysis was then carried out for the case study of a dynamic ORC power plant with 1 MW electricity output under Beijing's climate conditions. The costs of the power plant together with the composition tuning system were estimated according to published data and standard methods in literature. Compared to a conventional ORC power plant, the extra costs fall into two categories: the first being the extra volume of working fluid, and the second being the composition tuning system, essentially a small distillation system. The cost of the extra working fluid is estimated to be in the range of 6%. However the costs of the composition tuning system strongly depend the operating time required to complete the separation of the two components. For the worst case scenario, the whole charge needs to be separated in several hours. The required composition system in this case will add about 0.4% to the total cost of the power plant. This value starts to become negligible as the response time increases to over 100 h.

The results of the economic analysis predict that the dynamic cycle will have a shorter payback period and increased NPV and return on capital employed for any value of ψ greater than 8.2%, which corresponds to an annual temperature variation of 22 °C. For the case study of Beijing's climate conditions combined with a heat source temperature of 100 °C, the payback period of a dynamic ORC power plant is around 19% shorter than for the conventional system. This shows the proposed dynamic ORC concept has a great potential for the application to low temperature heat sources such as geothermal energy and industrial waste heat, under continental climate conditions.

Acknowledgement

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References

[1] UK Government: climate change act. <www.parliament.uk> [accessed 2008].
[2] Bianchi M, De Pascale A. Bottoming cycles for electric energy generation: parametric investigation of available and innovative solutions for the exploitation of low and medium temperature heat sources. Appl Energy 2011;88:1500–9.
[3] Yang K, Zhang H, Wang Z, Zhang J, Yang F, Wang E, et al. Study of zeotropic mixtures of ORC (organic Rankine cycle) under engine various operating conditions. Energy 2013;58:494–510.
[4] Franco A. Power production from a moderate temperature geothermal resource with regenerative Organic Rankine Cycles. Energy Sustain Dev 2011;15(4):411–9.
[5] Barbier E. Geothermal energy technology and current status: an overview. Renew Sustain Energy Rev 2002;6:3–65.
[6] Chamorro CR, García-Cuesta JL, Mondéjar ME, Pérez-Madrazo A. Enhanced geothermal systems in Europe: an estimation and comparison of the technical and sustainable potentials. Energy 2014;65:250–63.
[7] Habka M, Ajib S. Evaluation of mixtures performances in Organic Rankine Cycle when utilizing the geothermal water with and without cogeneration. Appl Energy 2015;154:567–76.
[8] Liu Q, Shen A, Duan Y. Parametric optimization and performance analyses of geothermal organic Rankine cycles using R600a/R601a mixtures as working fluids. Appl Energy 2015;148:410–20.
[9] Wang EH, Zhang HG, Fan BY, Ouyang MG, Zhao Y, Mu QH. Study of working fluid selection of organic Rankine cycle (ORC) for engine waste heat recovery. Energy 2011;36:3406–18.
[10] Zhang HG, Wang EH, Fan BY. A performance analysis of a novel system of a dual loop bottoming organic Rankine cycle (ORC) with a light-duty diesel engine. Appl Energy 2013;102:1504–13.
[11] Chen Q, Xu J, Chen H. A new design method for Organic Rankine Cycles with constraint of inlet and outlet heat carrier fluid temperatures coupling with the heat source. Appl Energy 2012;98:562–73.
[12] Yu C, Xu J, Sun Y. Transcritical pressure Organic Rankine Cycle (ORC) analysis based on the integrated-average temperature difference in evaporators. Appl Therm Eng 2015;88:2–13.
[13] Yue C, Han D, Pu W, He W. Thermal matching performance of a geothermal ORC system using zeotropic working fluids. Renew Energy 2015;80:746–54.
[14] Aghahosseini S, Dincer I. Comparative performance analysis of low-temperature Organic Rankine Cycle (ORC) using pure and zeotropic working fluids. Appl Therm Eng 2013;54:35–42.
[15] Chen H, Goswami Y, Rahman MM, Stefanakos EK. A supercritical Rankine cycle using zeotropic mixture working fluids for the conversion of low-grade heat into power. Energy 2011;36:549–55.
[16] Lecompte S, Ameel B, Ziviani D, van den Broek M, De Paepe M. Exergy analysis of zeotropic mixtures as working fluids in Organic Rankine Cycles. Energy Convers Manage 2014;85:727–39.
[17] Braimakis K, Preißinger M, Brüggemann D, Karellas S, Panopoulos K. Low grade waste heat recovery with subcritical and supercritical Organic Rankine Cycle based on natural refrigerants and their binary mixtures. Energy 2015;88:80–92.
[18] Xu J, Yu C. Critical temperature criterion for selection of working fluids for subcritical pressure Organic Rankine Cycles. Energy 2014;74:719–33.
[19] Harinck J, Calderazzi L, Colonna P, Polderman H. ORC deployment opportunities in gas plants. In: 3rd International seminar on ORC power systems. Brussels, Belgium; 2015.
[20] TuTiempo.net. Climate XIAN; 2013. <http://en.tutiempo.net/climate/2013/ws-570360.html>.
[21] Lemmon EW, Huber ML, McLinden MO. NIST standard reference database 23: reference fluid thermodynamic and transport properties-REFPROP, version 9.1. Gaithersburg: National Institute of Standards and Technology, Standard Reference Data Program; 2013.
[22] Abadi G, Yun E, Kim K. Flow boiling characteristics of R134a and R245fa mixtures in a vertical circular tube. Exp Therm Fluid Sci 2016;72:112–24.
[23] Liu Q, Duan Y, Yang Z. Effect of condensation temperature glide on the performance of Organic Rankine Cycles with zeotropic mixture working fluids. Appl Energy 2014;115:394–404.
[24] Gawlik K, Hassani V. Advanced binary cycles: optimum working fluids. In: Energy conversion engineering conference. vol. 3. Honolulu, HI, USA; 1997. p. 1809–14.
[25] Venkatarathnam G, Murthy S. Effect of mixture composition on the formation of pinch points in condensers and evaporators for zeotropic refrigerant mixtures. Int J Refrig 1999;22:205–15.
[26] Wang XD, Zhao L. Analysis of zeotropic mixtures used in low-temperature solar Rankine cycles for power generation. Sol Energy 2009;83:605–13.
[27] Rothfleisch P. A simple method of composition shifting with a distillation column for a heat pump employing a zeotropic refrigerant mixture. Maryland, USA: Technical Report of NIST; 1995.
[28] Pierobon L, Nguyen TV, Mazzucco A, Larsen U, Haglind F. Part-load performance of a wet indirectly-fired gas turbine integrated with an Organic Rankine Cycle turbogenerator. Energies 2014;7(12):8294–316.
[29] Clemente S, Micheli D, Reini M, Taccani R. Simulation model of an experimental small-scale ORC cogenerator. In: ORC 2011 first international seminar on ORC power systems. TU Delft, The Netherlands; 2011.
[30] Collings P, Yu Z. Modelling and analysis of a small-scale Organic Rankine Cycle system with a scroll expander. In: Proceedings of the world congress on engineering; 2014. p. 2.
[31] Wang XQ, Li XP, Li YR, Wu CM. Payback period estimation and parameter optimization of subcritical Organic Rankine Cycle for waste heat recovery. Energy 2015;88:734–45.
[32] Kang S. Design and experimental study of ORC (organic Rankine cycle) and radial turbine using R245fa working fluid. Energy 2012;41(1):514–24.
[33] Gorak A, Sorensen E. Distillation: fundamentals and principles. London: Academic Press; 2014.
[34] UNEP Ozone Secretariat: 1998 report of the refrigeration, air conditioners and heat pumps technical options committee; 1998.
[35] Stichlmair J, Stemmer A. Reduction of energy requirements in distillation. Chem Eng Technol 1989;12:163–9.
[36] Zhang S, Wang H, Guo T. Performance comparison and parametric optimization of subcritical Organic Rankine Cycle (ORC) and transcritical power cycle system for low-temperature geothermal power generation. Appl Energy 2011;88:2740–54.
[37] Quoilin S, Broek M, Declaye S, Dewallef P, Lemort V. Techno-economic survey of Organic Rankine Cycle (ORC). Renew Sustain Energy Rev 2013;22:168–86.
[38] Staffell I, Green R. How does wind farm performance decline with age. Renew Energy 2014;66:775–86.
[39] Stougie L, Kooi H. Possibilities and consequences of the total cumulative exergy loss method in improving the sustainability of power generation. Energy Convers Manage 2016;107:60–6.
[40] Lecompte S, Lemmens S, Huisseune H, Broek M, Paepe M. Multi-objective thermo-economic optimization strategy for ORCs applied to subcritical and transcritical cycles for waste heat recovery. Energies 2015;8:2714–41.